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AUTOMOTIVE FUEL ECONOMY--POTENTIAL IMPROVEMENT THROUGH SELECTED ENGINE AND DIFFERENTIAL GEAR LUBRICANTS

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FINAL REPORT

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16. Abstract

This report evaluates the effects of four engine lubricants and three differential gear lubricants on the fuel economy of two 1978 automobiles operated at 20°, 70°, and 100° F ambient temperatures.

The engine lubricants were evaluated using the 1978 Federal Test procedure and steady state tests from a cold start. The gear lubricants were evaluated in steady state operation from a cold start.

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PREFACE

This report, prepared by the U.S. Department of Energy, Bartlesville Energy Technology Center, Bartlesville, Oklahoma; for the U.S. Department of Transportation, Transportation Systems Center, Energy Technology Branch, Cambridge, MA, describes experimental work and findings in a study of fuel economy effects of four engine lubricants and three differential gear lubricants on two 1978 automobiles operated over the temperature range encountered in the United States using the Federal test procedure and 60-mph steady state tests.

This project is funded by the National Highway Traffic Safety Administration, Office of Research and Development, Office of Passenger Vehicle Research, Technology Assessment Division under PPA HS153.

Mr. James A. Kidd, Jr. of the U.S. Department of Transportation, Transportation Systems Center, is the technical monitor.

METRIC CONVERSION FACTORS

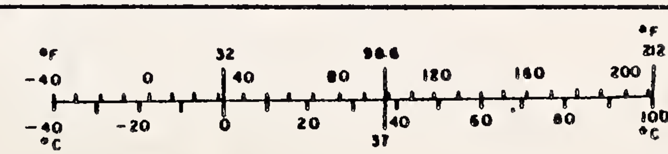
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.46	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. INTRODUCTION

Over the past few years, passenger car fuel economy has become a major issue in the automotive industry and the various regulatory and technical governmental agencies. With the skyrocketing cost of a barrel of crude oil and the need to reduce our dependence on foreign oil producers, more fuel efficient automobiles must be manufactured to achieve the federally mandated fuel economy standards. While fuel economy is a very complex function of many variables, it is essentially a function of seven factors:[1*] friction between moving parts in a vehicle; aerodynamic drag; tire rolling resistance; vehicular weight; driving characteristics; ambient temperature; and trip length. This report discusses two of these factors: friction reduction via selected engine and differential gear lubricants, and the magnitude of their effects at three ambient temperatures (20°, 70°, and 100°F).

In a spark-ignition, internal-combustion engine, less than 30 percent of the energy supplied by the fuel is converted to mechanical work, and approximately 70 percent is lost to the exhaust, coolant, and lubricant.[2] Of the 30 percent of the fuel energy available to do mechanical work, about 12 to 25 percent is translated to brake horsepower at the drive wheels. Approximately 5 to 8 percent is spent in overcoming losses in the transmission and drive lines. At wide open throttle, engine friction losses account for approximately 5 percent of the energy provided by the fuel.[2] At part throttle, friction losses account for as much as 8 percent of the fuel energy converted to do mechanical work. This shows the potential improvement in engine efficiency through low viscosity and friction-modified engine oils and suggests the operating range in which the most dramatic improvement in fuel economy might be expected. Even small

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* Superscript numbers designate references at end of report.

incremental improvements in fuel economy on the order of 1 to 5 percent can be significant on a large scale, especially considering that the average American car burns more than its own weight in gasoline each year--about 700 gallons totaling over 28 percent of petroleum consumption in the United States.[3]

Most passenger car differentials employ hypoid gears with a spiral bevel design.* Hypoid gears typically operate between 86 and 98 percent efficiency at high loads.[4] At lighter loads, the efficiency is somewhat less, about 70 percent under city driving conditions.

Power losses through the differential are mainly due to churning (work required to overcome internal fluid friction), friction between moving parts such as gear teeth, and vibration from various sources.[5]

In most cases, friction and churning losses account for most of the power losses at high and low loads. Friction losses manifest themselves in heat losses and higher operating temperatures which result in lower axle efficiency and shortened lubricant life.[5]

Relative fuel savings of up to 5 percent have been reported in short trip service with lower viscosity gear lubricants.[5] In general, the lower the viscosity of the gear lubricant, the higher the axle efficiency. However, if the viscosity of the lubricant is inadequate to maintain hydrodynamic lubrication at peak loads and extreme temperatures, metal-to-metal contact begins, and the resulting increased operating temperature of the lubricant could lead to boundary lubrication which is characterized by increased power losses, wear, and reduced efficiency.[6]

*Gears with a spiral bevel design have curved teeth and operate more quietly than those with straight teeth, because the curved teeth make a sliding contact. Also, the spiral bevel ring gear and pinion arrangement is stronger than a straight tooth setup, because more than one tooth is in contact at all times.

Hypoid gears have the spiral bevel design, but the center of the pinion shaft is below the center of the ring gear. The advantage of this design is that it allows the drive shaft to be placed lower, thus reducing the floor hump.

2. EXPERIMENTAL EQUIPMENT AND PROCEDURES

2.1 VEHICLES

Two vehicles were used in this program:

1. A 1978 Buick Century equipped with a 231-CID (3.8L), V6, 2-bbl engine (used in both engine and gear lubricants studies).

2. A 1978 Ford Fairmont equipped with a 140-CID (2.3L), 4-cylinder, 2-bbl engine (used in the engine lubricants study).

A detailed description of the two vehicles is given in Table 1. The vehicles were purchased new and accumulated approximately 4,000 miles of city/highway driving to stablize exhaust emissions and fuel economy

TABLE 1. TEST VEHICLE DESCRIPTION

DOE/BETC ID No......	179	192
Manufacturer........	Buick	Ford
Model...............	Century	Fairmont
Engine..............	V6	4-cyl
Displacement, CID...	231	140
Carburetion.........	2-bbl	2-bbl
Compression ratio...	8.0:1	9.0:1
Horsepower:		
SAE Net...........	105	89
@ RPM.............	3400	4800
Transmission........	Auto	Auto
Axle ratio..........	2.73	3.08
Air conditioning....	yes	yes
Power steering......	yes	yes
Power brakes........	yes	yes
Curb wt, lb.........	3158	2748
Inertia wt, lb......	3500	3000
Emission controls:		
AIR...............	no	yes
Catalyst..........	yes	yes
EGR...............	yes	yes
Carbon canister...	yes	yes

2.2 FUELS

The test fuel was a summer grade fuel with 10 psi Reid vapor pressure,* 0.718 specific gravity, and 50 percent distillation at 218°F.

2.3 ENGINE LUBRICANTS

Table 2 describes the properties of the following test lubricants:

1. A commercial 10W40 premium quality SE (base lubricant).
2. An experimental 5W20 synthetic SE/CC.**
3. A commercial 10W40 premium quality synthetic SE/CC.
4. A commercial 10W40 premium SE/CC containing graphite in colloidal suspension.

2.4 GEAR LUBRICANTS

The following lubricants were used in this study:

1. A commercial multigrade 85W90 (base lubricant).
2. A commercial 75W synthetic
3. An experimental 75W mineral containing one percent molybdenum disulfide (MoS_2).

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* Reid vapor pressure (Rvp) is the gage vapor pressure of 38°C (100°F) gasoline in a rigid container having a four to one volumetric ratio of air to liquid.

** SE = The second highest (fifth) American Petroleum Institute service classification for spark ignition engine oil. SF is the highest classification.

CC = The second highest (third) API service classification for compression ignition engine oil.

TABLE 2. ENGINE AND GEAR LUBRICANTS' PROPERTIES

	Engine Lubricants				Gear Lubricants		
	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite	85W90 Base	75W Synthetic	75W w/MoS_2
Flash point, °F.....	415	455	460	420	430	350	365
Kinematic viscosity:							
CS @ 40° C........	85.10	61.90	93.70	85	181.5	29.76	31.56
CS @ 100° C.......	14.28	6.87	14.40	15.16	15.22	5.25	5.95
Viscosity index.....	142	65	138	147	95	120	136
Sulfated ash,							
pct wt............	0.72	1	0.79	0.75	0.04	0.04	0.07
Elemental analysis,							
pct wt:							
Ba................	0.03	0.03	0.03	0.03	Trace	Trace	0.04
Ca................	0.03	0.03	0.02	0.03	Trace	0.002	Trace
Mg................	0.005	0.002	0.005	0.005	Trace	0.004	Trace
Zn................	0.006	0.006	0.006	0.006	0.003	0.005	0.006
P.................	0.15	0.27	0.23	0.21	Trace	Trace	Trace
S.................	0.34	0.37	0.44	0.57	1.60	1.79	1.34
API service.........	SE	SE/CC	SE/CC	SE/CC	GL5	GL5	GL5

2.5 DRIVING CYCLES

Two different cycles were used in this study:

1. The 1978 Federal test procedure (FTP), five-bag test, was used in the engine lubricants study.[7]

2. Sixty mph, steady state from cold start was used in both the engine and gear lubricants study.

The FTP conformed to the Code of Federal Regulations.[7] (See Figure 1.)

The steady-state test consisted of a cold start after an overnight soak at the test ambient temperature, followed immediately by acceleration to 60 mph according to the following equation:

$$v = 1.5t$$

where,

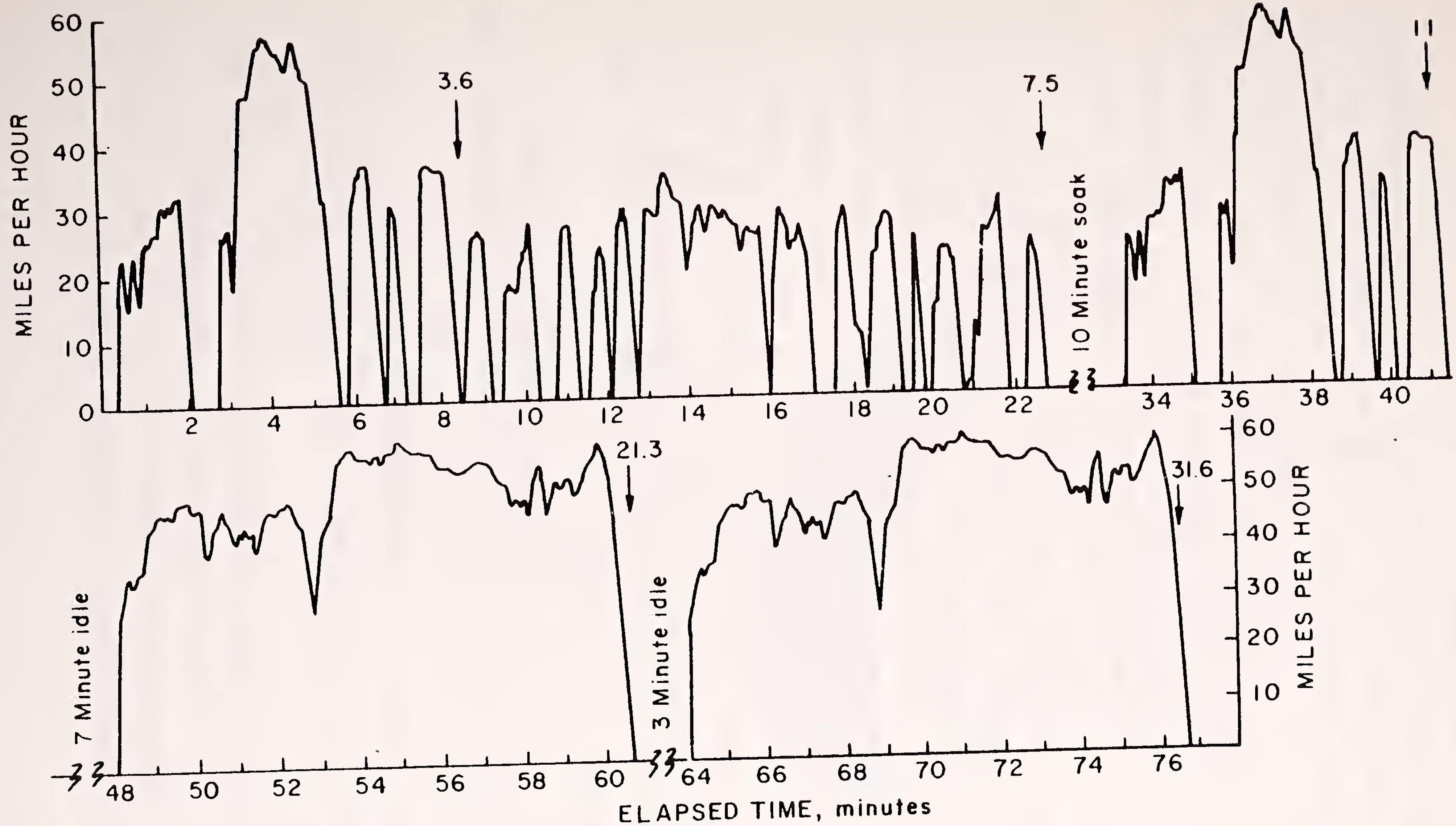
v = vehicular speed, miles per hour

t = elapsed time from cold start, seconds

The acceleration time from 0 to 60 mph was 40 seconds. Once the vehicle reached 60 mph, the driver maintained that speed for 45 minutes. The total distance traveled was 45 miles. The driver followed a prescribed driving schedule during each test, and fuel economy was calculated every 60 seconds by the carbon balance method using a constant volume sampling system and a computerized data acquisition system.

2.6 AMBIENT TEST TEMPERATURES

All tests were conducted at 20°, 70°, and 100°F ambients. In the case of the 20°F ambient, it was necessary, in some instances, to start the test near 15°F to obtain an average test temperature near 20°F. Triplicate tests were conducted at each ambient temperature for each engine lubricant, and duplicate tests were run for each differential gear lubricant. The results presented in Appendices A, B, and C are based on the average of the triplicate and duplicate tests.



Note: Numerical Values with Arros Toward x Axis Indicate Miles at End of Each Test Segment.

FIGURE 1. 1978 FTP DRIVING SCHEDULE

2.7 INSTRUMENTS AND APPARATUS

The vehicle tests were conducted on a chassis dynamometer equipped with road load power control and direct-drive inertia system in a climate-controlled test cell. Ambient temperature capabilities were 15° to 100°F. The exhaust gas analytical system consisted of a flame ionization detector, for determination of hydrocarbon; nondispersive infrared analyzers, for determination of carbon monoxide and carbon dioxide; and a chemiluminescence analyzer, for determination of nitrogen oxide. The exhaust emissions were collected using the "bag sampling" technique, and fuel economy was calculated using the carbon balance method. In addition, to properly evaluate the effects of the gear lubricants on axle horsepower losses and efficiency, the test vehicle was outfitted with a wheel torque sensor and a driveshaft torque sensor. Both torque sensors were outfilled with speed sensors which provided analog signals proportional to wheel speed. An exploded view of the wheel torque sensor is shown in Figure 2.

The wheel torque sensor assembly consisted of the following:

1. A wheel adapter bolted to the right rear brake drum of the vehicle to convert the bolt circle diameter from 4.75 inches to 5 inches.

2. An electrical resistance strain gage bridge built in a foil grid that was bonded to a disc and deformed in the same manner as the disc.

3. A second adapter to accommodate a slip-ring assembly.

4. The wheel and tire assembly that mounted on the first adapter.

5. A slip-ring assembly that bolted to the wheel and connected the torque signal back to an instrument.

The wheel torque sensor assembly moved the wheel approximately 2 inches outward from its original position.

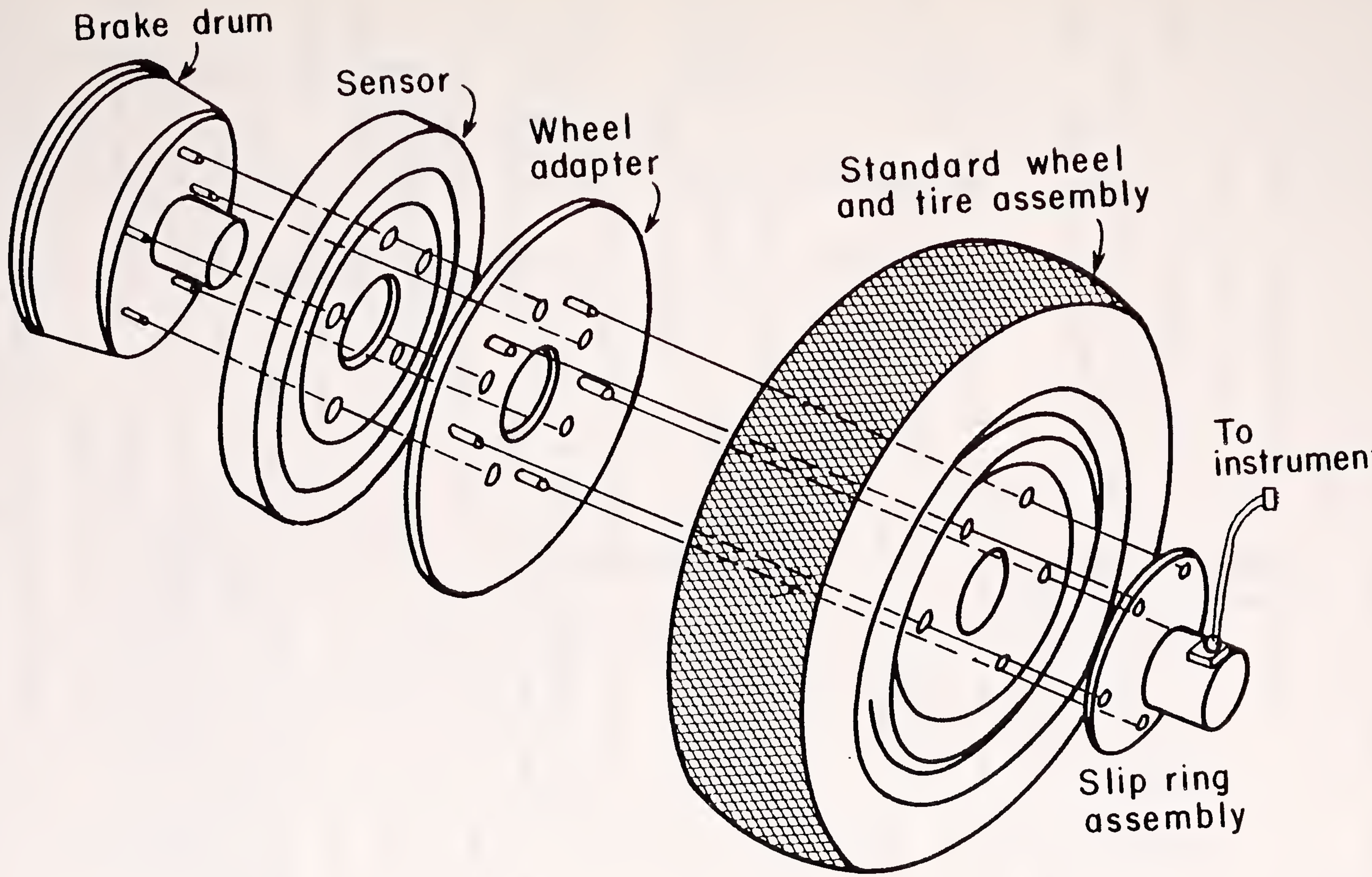


FIGURE 2. AUTOMOBILE WHEEL TORQUE SENSOR

2.8 TORQUE AND HORSEPOWER CALCULATION

The strain gage bridges and 60-tooth gears provide torque and revolutions per minute (RPM) readouts for the driveshaft and right rear wheel. In addition to empirical determination, the rear wheel RPM can be calculated by dividing the driveshaft RPM by the rear axle gear ratio (2.73:1). This RPM should be the same for both rear wheels unless slippage occurred or the vehicle began to turn. Neither was the case, since the vehicle was anchored when tested on the chassis dynamometer, and there was no slippage during acceleration.

The driveshaft torque sensor was essentially an in-line sensor with a four-arm bonded foil strain gage bridge. The non-rotating section was flanged on both sides and bolted to the driveshaft.

Torque measurement was based on torsional windup (angular deflection or strain) of a rotating shaft. The amount of angular deflection is always very small (0.5 to 1 degree typical).[8]

When torque was applied to the drive wheels of the vehicle through the rear axle, the axle shaft was twisted by loading. The wheel torque sensor provided an output voltage directly proportional to the applied torque. The strain gage bridge was connected to four silver slip rigns mounted on the rotating shaft. Silver graphite brushes rubbed on these slip rings and provided a path for the incoming excitation voltage and the output signal voltage. This signal was displayed by a digital indicator and then fed into a computerized data acquisition system. Torque, RPM, and horsepower were then measured and calculated every 60 seconds for 45 minutes. Torque for acceleration conditions was calculated as follows:

In general, in any rotating system (Figure 3), the sums of all torques must equal the product of the moment of inertia about the rotating axis and the angular acceleration:[8]

$$T_T - T_L = I_\alpha$$

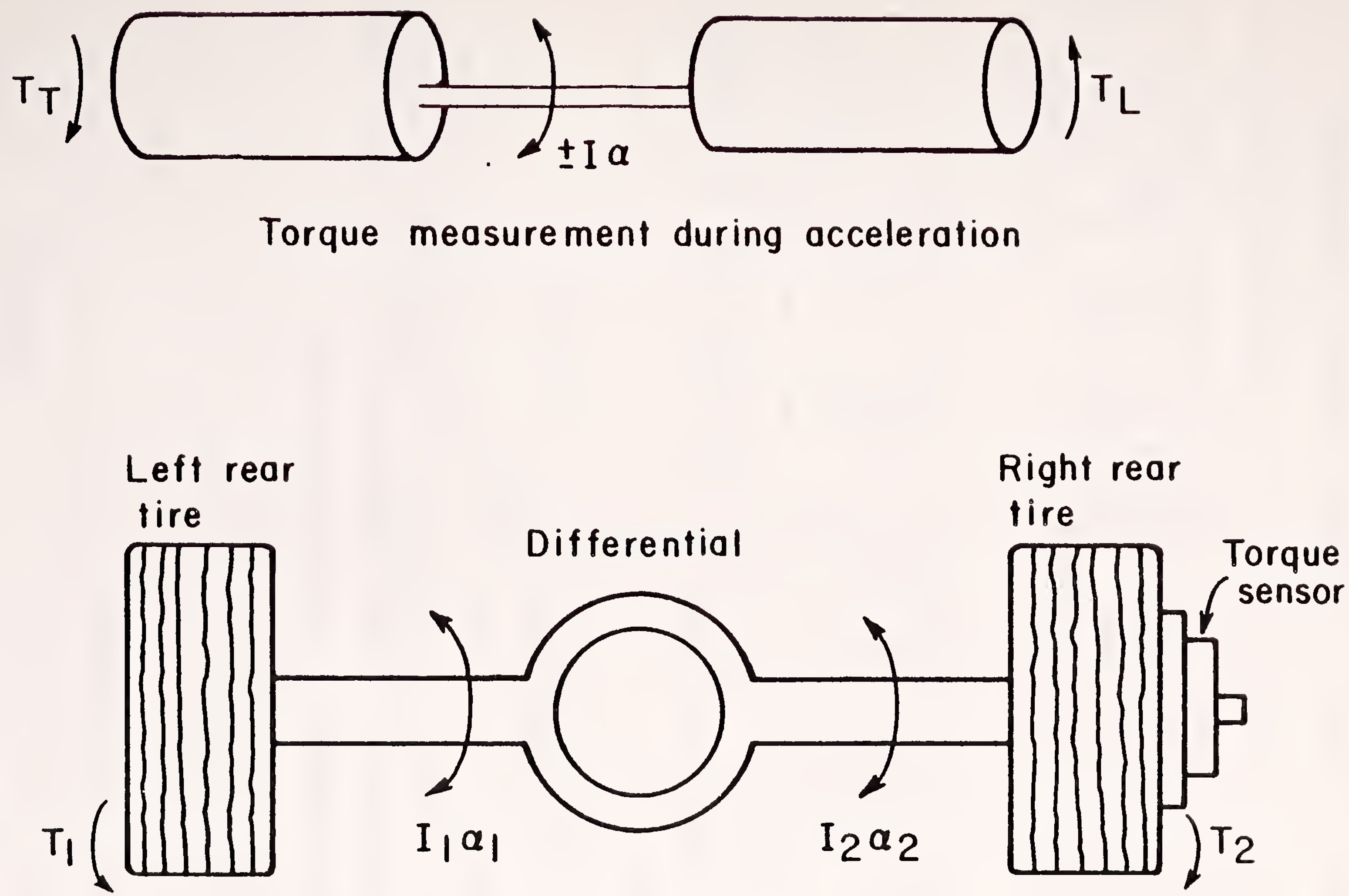


FIGURE 3. REAR VIEW OF AN AUTOMOBILE REAR AXLE UNDER ACCELERATING CONDITIONS

where,

T_T = Output torque of the test unit

T_L = Torque of the loading device

I = Moment of inertia of all rotating components

α = Angular acceleration of all rotating components

This equation points out that the torque of the test device and the torque of the loading device are only equal when the angular acceleration is zero. When the shaft is being accelerated, the torque of the test device will be greater than that of the load, and the converse will occur during deceleration.

In the case of an automobile rear axle (Figure 3) at rest, all the external forces and their moments about the axle shaft are balanced. During acceleration, these forces and their moments become unbalanced, and torque causes an angular acceleration about the axle shaft. The summation of torque must equal the product of the moment of inertia about the axle shaft times the angular acceleration:

$$T_1 + T_2 = I_1\alpha_1 + I_2\ {}_2 \qquad (1)$$

If the vehicle is at reat, then the initial angular velocity of the wheels is zero, and the initial angular displacement is zero. Then the general formula for rotation of a body about an axis through the center of gravity applies:

$$\theta = 1/2\ \alpha t^2$$

Where,

θ = Angular displacement

α = Angular acceleration

t = Elapsed time

Then:

$$W = \frac{d\theta}{dt} = \alpha t$$

This equation shows that the angular velocity (RPM) is directly proportional to angular acceleration. Therefore, unless the left and right rear wheels are turning at different speeds, the angular acceleration of both wheels are equal, and equation (1) can be rewritten:

$$T_1 + T_2 = (I_1 + I_2)\ \alpha$$

It should be mentioned here that the moment of inertia of the right rear wheel/tire assembly is higher than the left assembly because of the additional mass of the adapters, torque sensor, and slipring.

Since torque was measured only at the right rear wheel, it was then safe to multiply that reading by 2, only after the vehicle had reached 60 mph under cruising conditions and zero angular acceleration:

$$\alpha = \frac{dw}{dt} = o \text{ if } w = \text{constant}$$

Horsepower was calculated as follows:

$$\text{HP input} = \frac{\text{Torque Input, ft-lb} \times \text{RPM Driveshaft}}{5252}$$

$$\text{HP output} = \frac{\text{Torque Output ft-lb} \times \text{RPM Wheel}}{5252}$$

Horsepower losses were the difference between horsepower input and horsepower output. These losses were, as discussed previously in this report, the result of churning and friction in the differential.

The axle efficiency was calculated as follows:

$$\text{efficiency} = \frac{\text{HP Output}}{\text{HP Input}}$$

3. ENGINE LUBRICANTS - FTP EVALUATION

In order to show the relative effects of the various test lubricants on fuel economy of the test vehicles, the following was established:

1. The commercial 10W40 engine lubricant was considered the base lubricant through the entire program.
2. The percent change in fuel economy was calculated as follows:

$$\text{percent change} = \frac{FE_T - FE_B}{FE_B} \times 100$$

where,

FE_T = Average fuel economy with the test lubricant, mpg

FE_B = Average fuel economy with the base lubricant, mpg

During each test, the following temperatures were measured: coolant, oil, transmission fluid, carburetor air intake, air to vehicle, and differential gear oil. For the purpose of this discussion, only the most pertinent temperatures are shown in Figure 4.

Based on Figure 4, the vehicle warm-up period (approximately 20 minutes) appeared to be during the cold transient and stabilized portions of the 1978 Federal Test Procedure. During that period, the coolant temperature appeared to stabilize first, followed by the engine oil, transmission fluid, and the gear oil. The fully warmed-up conditions appeared to be during the hot transient and highway segments of the FTP.

3.1 20°F AMBIENT

As shown in Figure 5, the 5W20 synthetic lubricant averaged from 1.4 to 1.7 percent increase in fuel economy when compared to the 10W20 base lubricant. The 10W40 synthetic lubricant

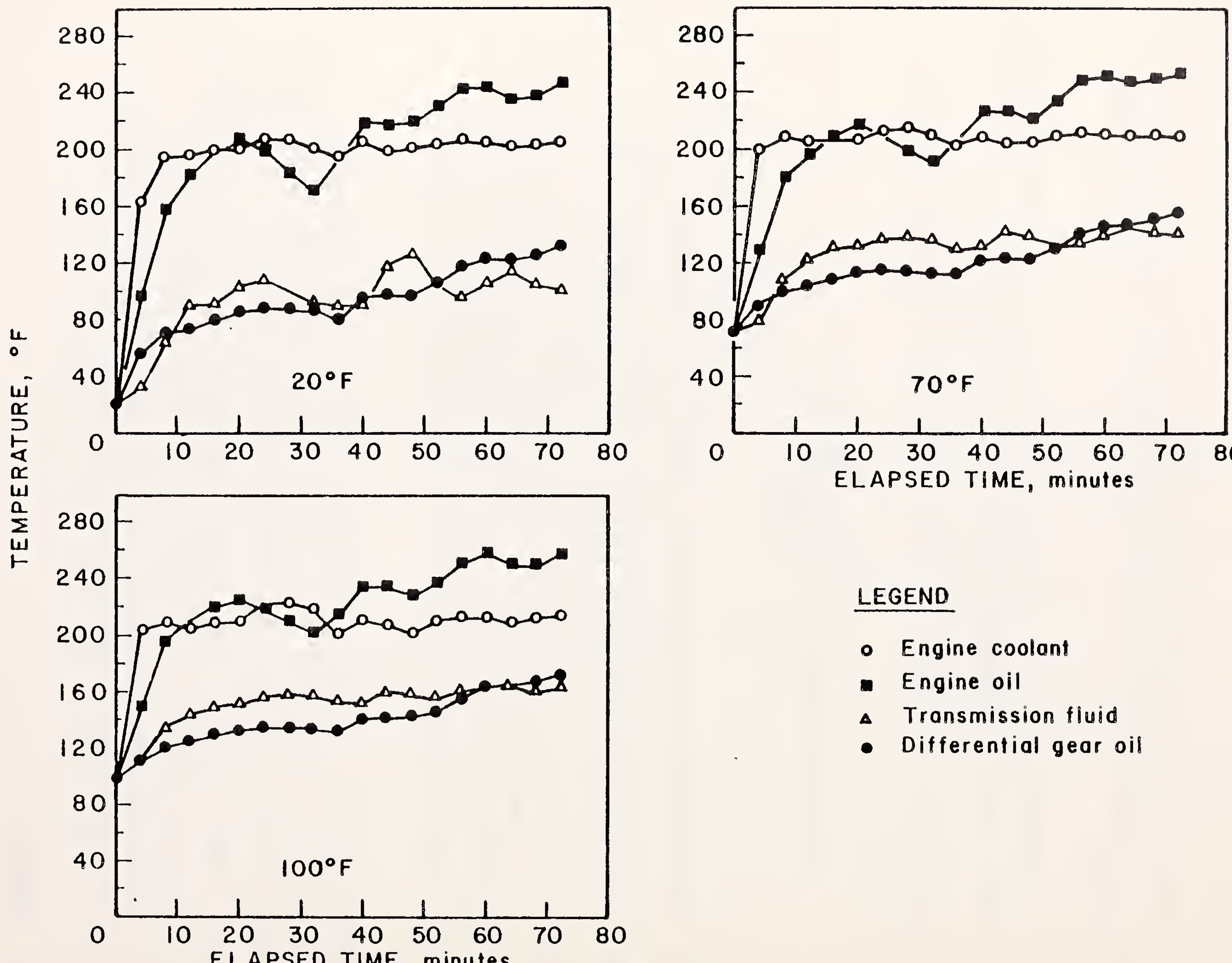


FIGURE 4. VEHICLE FLUID WARM-UP TEMPERATURES IN THE 1978 FEDERAL TEST PROCEDURE--140-CID FAIRMONT

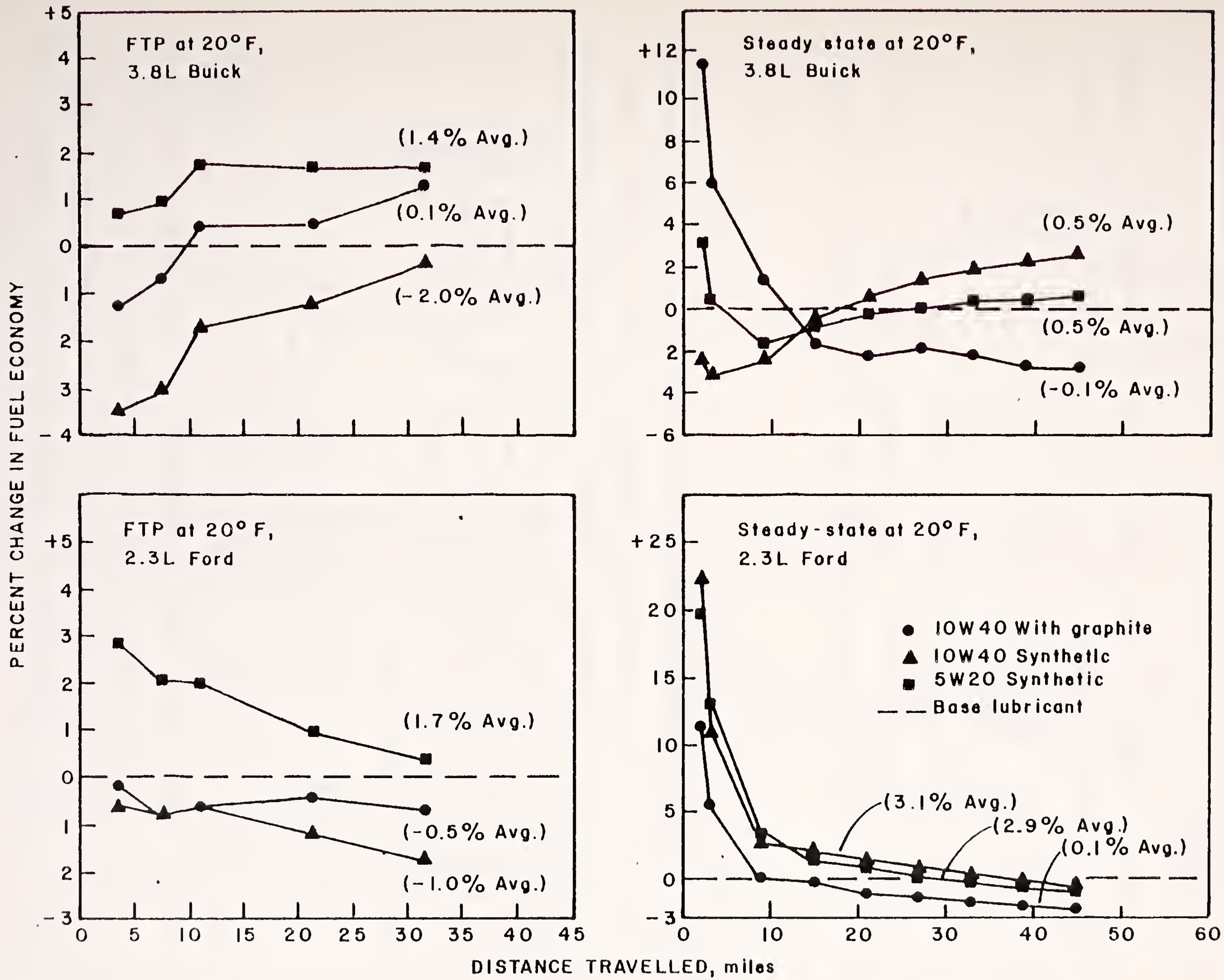


FIGURE 5. COMPARISON OF FUEL ECONOMY WITH TEST LUBRICANTS RELATIVE TO BASE LUBRICANT, 20°F

showed a consistent decrease in fuel economy throughout the test, averaging from 1.0 to 2.0 percent. The 10W40 graphite-containing lubricant showed essentially little or no change in fuel economy.

3.2 70°F AMBIENT

As shown in Figure 6, the 5W20 synthetic lubricant averaged from 1.1 to 3.5 percent increase in fuel economy when compared to the 10W40 base lubricant in all five segments of the FTP. The 10W40 synthetic lubricant showed a significant increase in fuel economy, ranging from 2.8 to 4.3 percent over the entire FTP. The 10W40 graphite lubricant averaged from 2.5 to 5.7 percent increase in fuel economy. The maximum benefit (7.6 percent) occurred in the Ford Fairmont during the cold-transient portion of the FTP.

3.3 100°F AMBIENT

At this ambient temperature, the 5W20 synthetic lubricant averaged from 0.2 to 0.7 percent increase in fuel economy when compared to the base lubricant (Figure 7). The 10W40 synthetic lubricant showed an average increase of 1.0 percent in fuel economy in one vehicle (Buick) and an average decrease of 1.3 percent in the second vehicle (Ford). The 10W40 graphite lubricant averaged from 0.1 to 1.4 percent increase in fuel economy over the entire cycle.

As illustrated in figures 5, 6, and 7, the fuel efficiency of the two test vehicles differed greatly from cold start to fully warmed-up conditions. It appeared that the percent change in fuel economy from the baseline of the 140-CID Ford was highest during warm-up from cold start and decreased as the vehicle was fully warmed-up. FTP warm-up data is in Figure 4 and Table 3. On the other hand, the fuel economy of the 231-CID Buick was lowest during warm-up from cold start and increased as the vehicle fluid tempterature stabilized. This implies that the frictional characteristics of the two engines, the fuel distribution, and the cold-start enrichment all play an important role in determining engine fuel efficiency under various conditions.

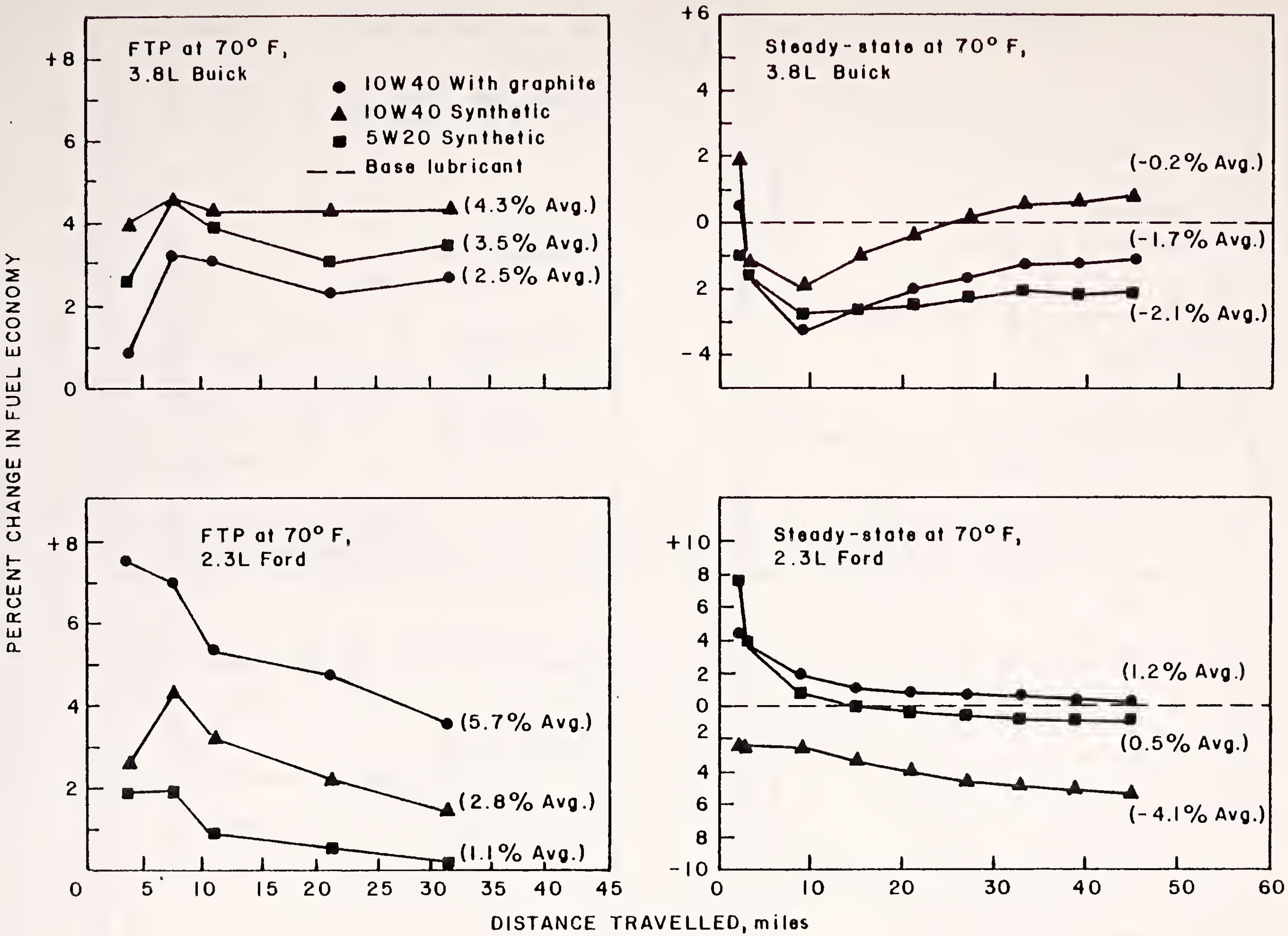


FIGURE 6. COMPARISON OF FUEL ECONOMY WITH TEST LUBRICANTS RELATIVE TO BASE LUBRICANT, 70°F

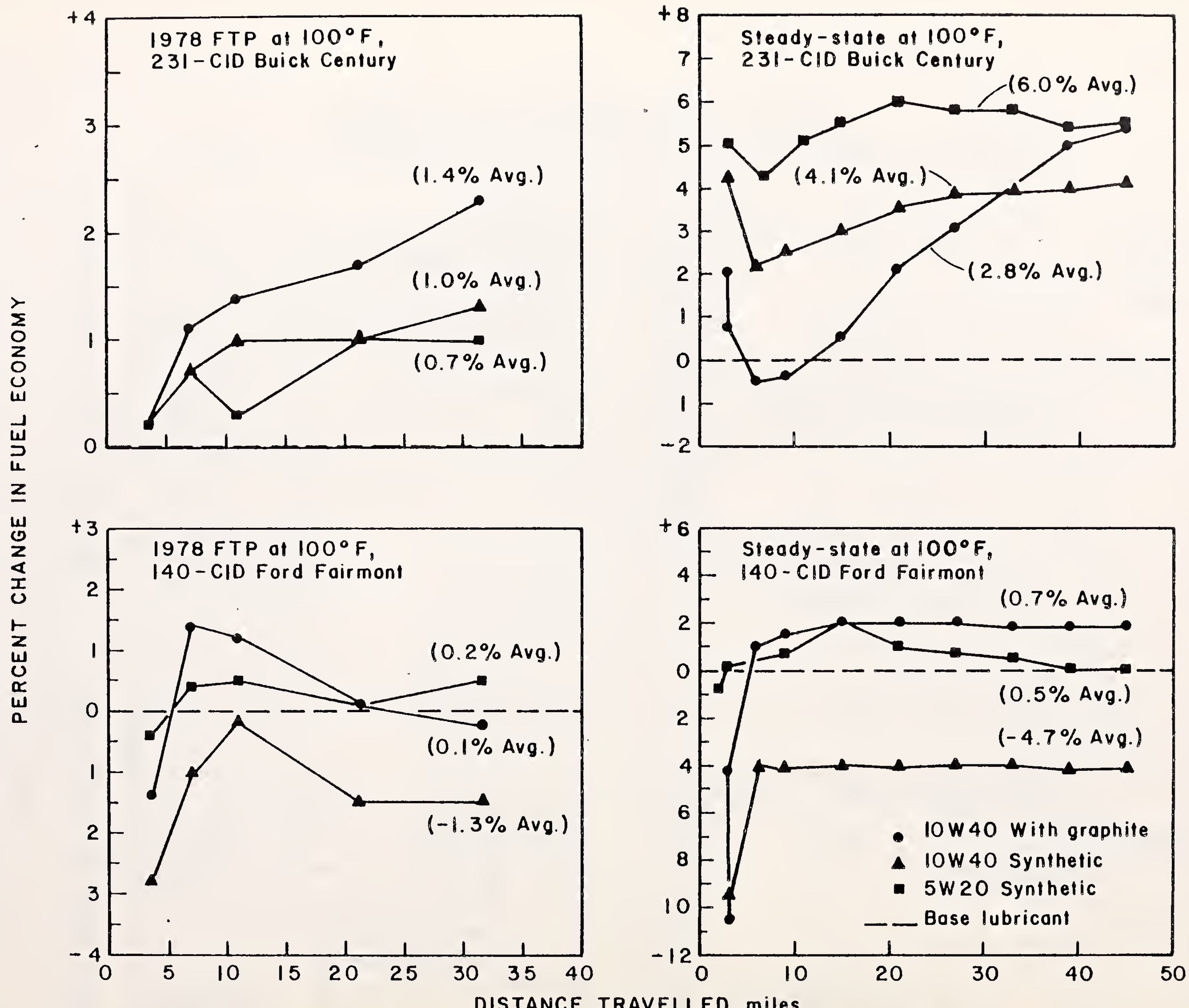


FIGURE 7. COMPARISON OF FUEL ECONOMY WITH TEST LUBRICANTS RELATIVE TO BASE LUBRICANT, 100°F

4. ENGINE LUBRICANTS COLD START TO 60 MPH STEADY STATE EVALUATION

Figure 8 shows the fluid temperatures during this test at the three ambient temperatures: 20°, 70°, and 100°F. Again, the engine coolant temperature appeared to stabilize first, followed by the oil, transmission fluid, and differential gear oil. It appeared that the differential gear oil temperature did not reach equilibrium until the distance traveled was near 40 miles. However, for the purpose of this discussion, it was established that 0-15 miles was the distance required for the vehicle to warm-up and 16-45 miles was the distance the vehicle traveled under fully warmed-up conditions.

4.1 20°F AMBIENT

As shown in Figure 5, the 5W20 synthetic lubricant averaged from 0.5 to 2.9 percent increase in fuel economy when compared to the base lubricant. The most dramatic increase occurred in the Ford immediately after start up (19.8 percent). The 10W40 synthetic lubricant averaged from 0.5 to 3.2 percent increase in fuel economy. Again, the most significant increase (22.5 percent) occurred in the Ford immediately after cold start. The 10W40 graphite lubricant showed no measurable change in fuel economy under fully warmed-up conditions. However, an increase in fuel economy (11.3 percent) occurred immediately after cold start in the Ford and the Buick (11.6 percent).

4.2 70°F AMBIENT

At this ambient temperature, the fuel economy with the three test lubricants was similar or slightly lower than with the 10W40 base lubricant (Figure 6). These findings were contrary to the results obtained in the FTP at the same ambient temperature, where all three test lubricants showed a marked increase in fuel economy over the baseline. However, during vehicle warm-up (0-15 miles), the 5W20 synthetic lubricant

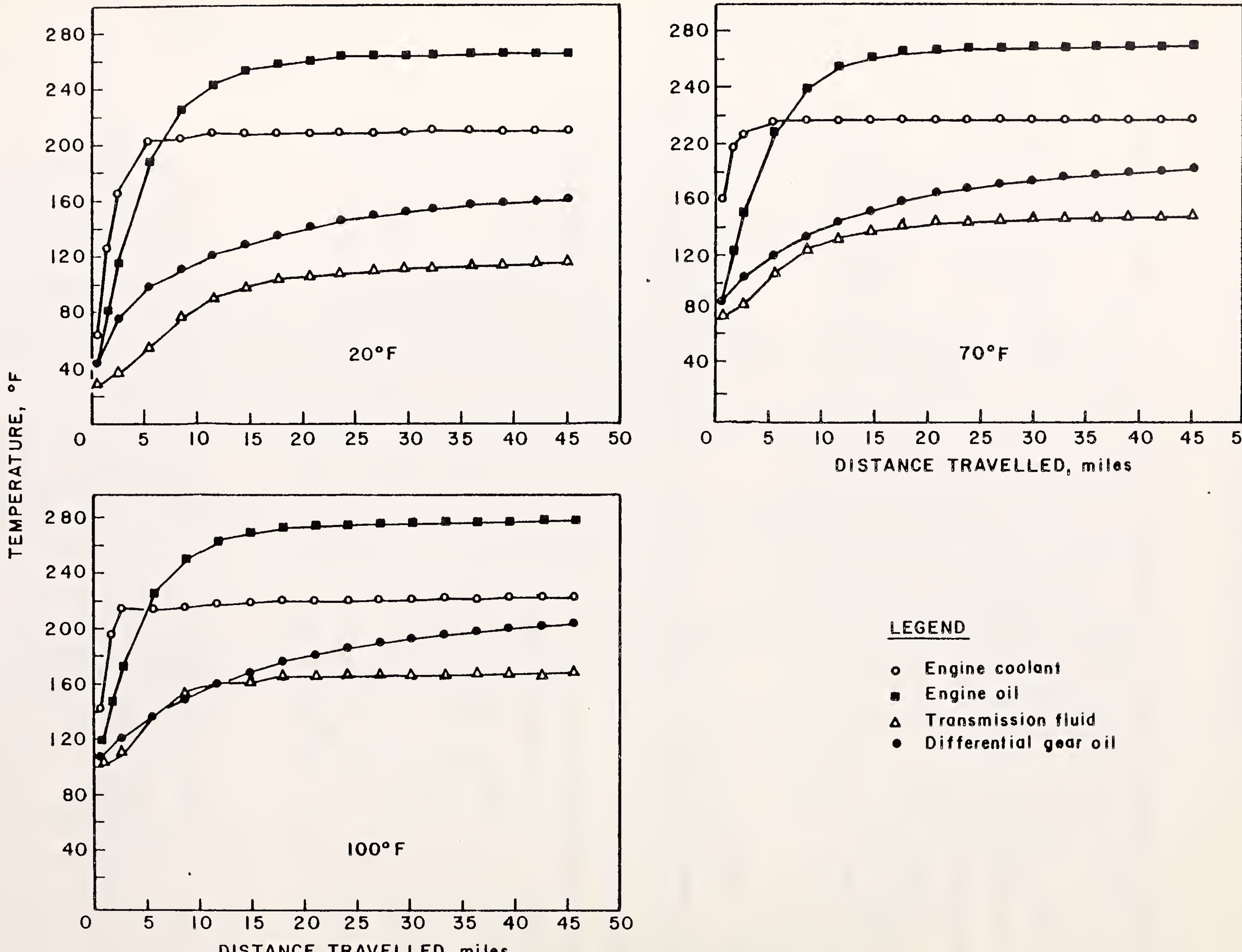


FIGURE 8. VEHICLE FLUID WARM-UP TEMPERATURES IN STEADY-STATE OPERATION--231-CID BUICK

and the 10W40 graphite lubricant averaged 3.7 and 3.9 percent increase in fuel economy with maximum increases of 7.7 and 4.5 percent immediately after cold start. This shows the influence of the test cycle on fuel economy benefits for both the low-viscosity and the friction-modified oils. Apparently, under light-load, low-speed operation such as in the FTP city cycle, fuel economy benefits with either low-viscosity or friction-modified oils are maximum; while, under light-load, high-speed operation such as in highway cruising, fuel economy benefits are minimum. This closely correlates with the fact that under part-throttle conditions, friction losses in an internal combustion engine account for as much as 8 percent of the fuel energy converted to do mechanical work, and suggests the operating range in which the most increase in fuel economy might be expected with low-viscosity or friction-modified oils.

4.3 100°F AMBIENT

The engine response to the test lubricants at this temperature varied significantly from one vehicle to the other (Figure 7). In the case of the Buick, the 5W20 and 10W40 synthetic lubricants averaged 6.0 and 4.1 percent increase in fuel economy from cold start to fully warmed-up conditions while the graphite lubricant averaged about 2.8 percent. In the case of the Ford, the 5W20 synthetic and 10W40 graphite lubricants averaged less than 1.0 percent increase in fuel economy, while the 10W40 synthetic lubricant showed a significant decrease of 4.7 percent when compared to the base lubricant.

Extrapolating from the data indicates that employing a 5W30, a 7.5W30, or a 10W30 synthetic or friction modified mineral oil probably will improve fuel economy.

5. OIL TEMPERATURES

Table 3 shows the oil temperatures for the four test lubricants in the two vehicles during the FTP, and Table 4 shows the oil temperatures in steady state operation. In the Buick, the temperature of the oil containing graphite consistently ranged from 15° to 30°F lower than the other test oils. This suggests a decrease in frictional losses and, correspondingly, an increase in engine efficiency and fuel economy. This certainly was the case with the Buick at the three ambient temperatures.

In the Ford, the graphite oil temperature was about the same as the temperatures of the other test oils. This suggests no reduction in frictional losses and no increase in fuel economy, but this was not the case in the FTP since there was an increase in fuel economy even though the oil temperature was the same.

TABLE 3. RELATIVE OIL TEMPERATURES IN THE FTP

Elapsed time, min.	20° F Ambient				70° F Ambient				100° F Ambient			
	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite
BUICK												
0	22	22	22	22	66	72	71	67	100	99	100	99
12	183	188	181	157	200	201	196	176	211	214	210	187
24	199	207	201	176	215	212	210	189	219	221	220	197
36	201	200	199	170	213	206	208	192	213	227	219	181
48	223	215	217	202	231	218	224	210	229	231	226	218
60	240	244	244	220	252	250	250	233	258	253	258	239
72	246	247	245	220	254	249	253	236	260	257	262	229
FORD												
0	20	19	20	19	70	63	65	74	100	97	98	97
12	150	146	149	147	171	171	174	174	188	187	187	188
24	154	155	156	154	181	181	180	181	197	195	196	195
36	160	153	126	157	181	186	190	191	197	198	197	205
48	173	164	171	167	193	196	195	196	207	207	207	209
60	174	183	188	185	209	206	205	205	220	217	220	217
72	184	186	189	187	211	209	204	204	214	217	224	216

TABLE 4. RELATIVE OIL TEMPERATURES IN STEADY-STATE OPERATION

Elapsed time, min.	20° F Ambient				70° F Ambient				100° F Ambient			
	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite	10W40 Base	5W20 Synthetic	10W40 Synthetic	10W40 w/Graphite
BUICK												
0	21	24	23	26	71	70	71	72	101	101	101	102
9	219	226	222	174	234	239	238	221	252	248	251	232
18	255	253	254	221	260	258	262	249	271	266	274	254
27	260	256	258	228	263	260	265	252	277	272	280	255
36	261	257	259	228	264	260	266	253	279	274	283	256
45	262	258	260	228	264	260	266	253	280	275	285	256
FORD												
0	18	19	18	21	70	67	68	69	98	97	100	97
9	167	182	177	180	194	195	197	198	212	206	215	212
18	194	199	194	197	218	213	218	217	228	220	230	225
27	199	202	199	200	220	215	219	220	230	224	233	228
36	201	201	202	204	221	215	219	219	230	225	234	230
45	204	202	203	204	220	215	219	219	229	225	234	229

6. EFFECT OF HOT START VS COLD START

Tests were conducted on the Buick at 70°F ambient to determine the effect of hot start vs cold start on fuel economy and vehicle fluid temperatures in steady state operation.

The vehicle was soaked overnight at 70°F in the chassis dynamometer test cell. The engine was started cold, and the vehicle was driven through the steady state cycle described previously. At the end of the cycle, the vehicle was allowed to soak for 10 minutes at the test temperature, and was then restarted hot and run on the same cycle for 45 minutes. The coolant, oil, transmission fluid, and differential gear oil temperatures were measured in both cases, and fuel economy was calculated by the carbon balance method.

The results are illustrated in Figures 9 and 10. Immediately after starting the engine and for the first 15 miles, the increase in fuel economy during the hot start over the cold start averaged about 20.8 percent. From 16 to 45 miles, the average increase was about 6.0 percent.

The vehicle fluid temperatures appeared to stabilize at a much faster rate when started hot. The differential gear oil temperature appeared to reach equilibrium toward 40 miles from cold start and almost immediately after hot start. The same applied to the oil, transmission fluid, and coolant temperatures.

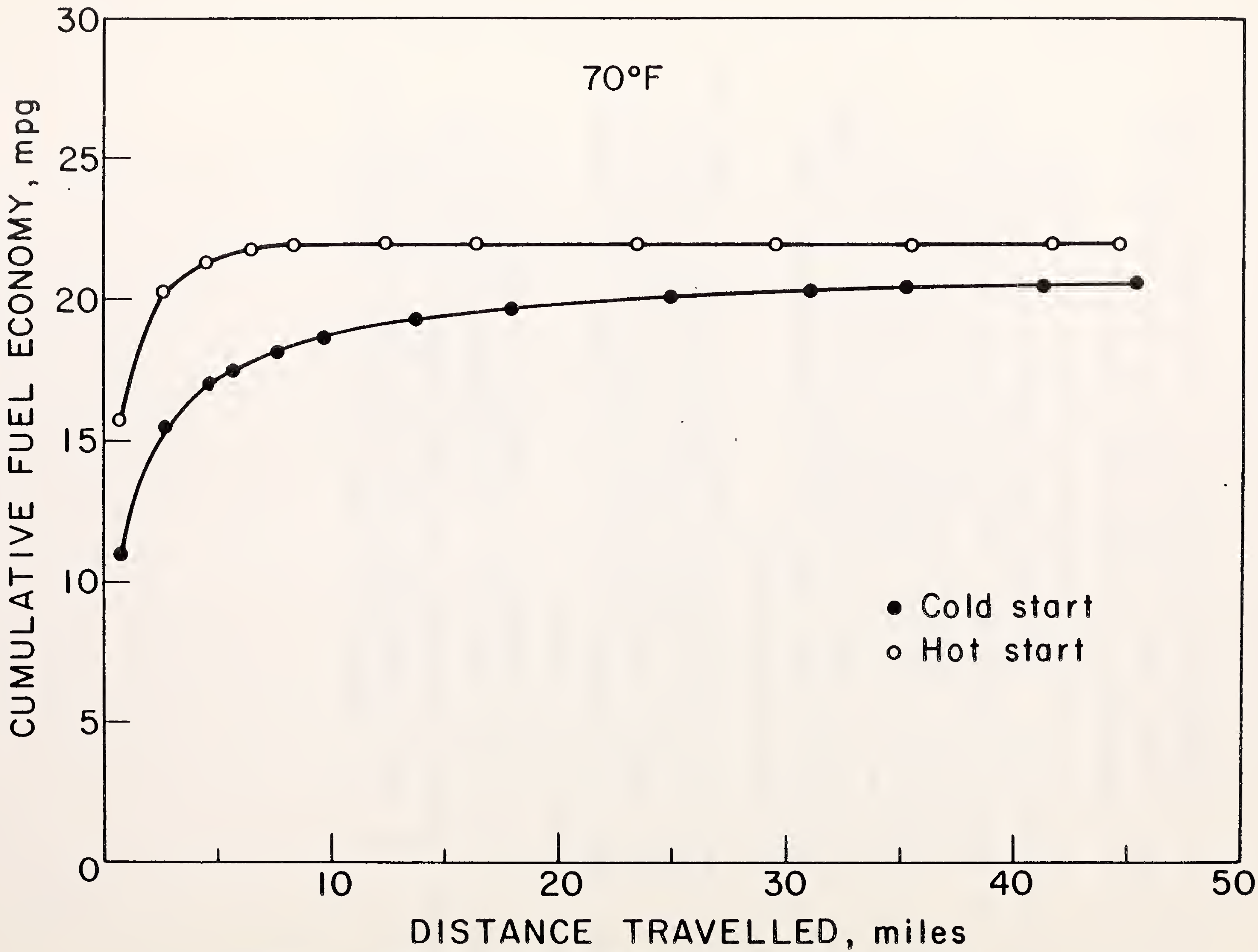


FIGURE 9. EFFECT OF COLD START VS HOT START ON VEHICLE FUEL ECONOMY, 70°F, 231-CID BUICK

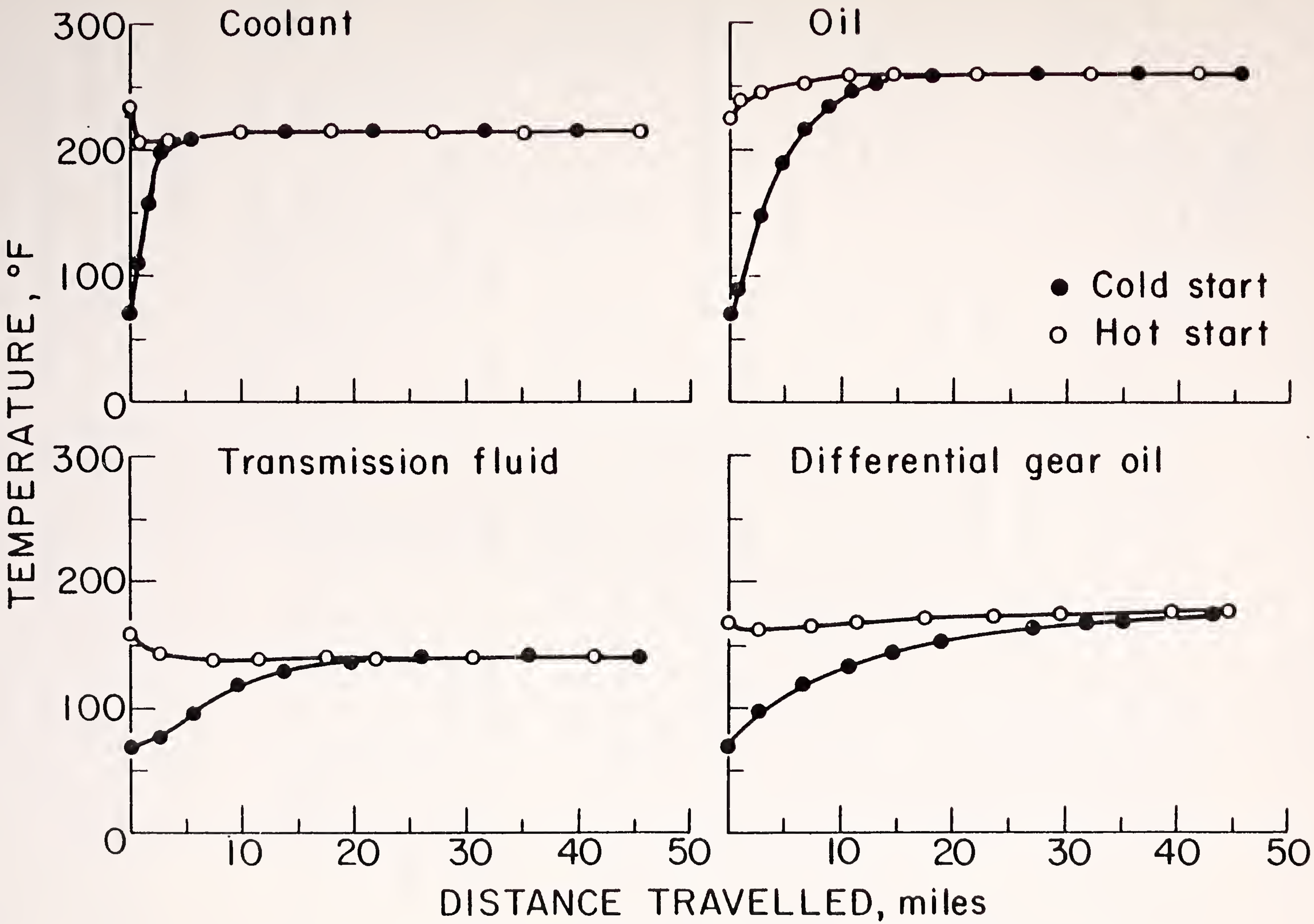


FIGURE 10. EFFECT OF HOT START VS COLD START ON VEHICLE FLUID TEMPERATURES, 70°F, 231-CID BUICK

7. DIFFERENTIAL GEAR LUBRICANTS - COLD START TO 60 MPH STEADY STATE EVALUATION

7.1 20°F AMBIENT

At this temperature, the 75W synthetic lubricant and the 75W lubricant containing MoS_2 showed significant increases in fuel economy over an 85W90 base lubricant only during vehicle warm-up from cold start (Figure 11). During vehicle warm-up (0-15 miles), the 75W synthetic lubricant averaged 9.7 percent increase in fuel economy when compared to the base 85W90. However, under fully warmed-up conditions, the 75W synthetic lubricant showed a slight decrease in fuel economy. These results were confirmed in Figure 12, which showed an average of 4.5 hp loss for the 75W synthetic lubricant during warm-up and 5.5 hp loss for the 85W90 base lubricant. However, under fully warmed-up conditions, the 75W synthetic lubricant showed a higher average horsepower loss (3.2 hp) than the 85W90 (2.2 hp). These results applied to the rear axle efficiency as well, which showed a higher efficiency for the 75W synthetic lubricant during warm-up (83.5 vs 81.0 percent) and a lower efficiency (87.7 vs 92.2 percent) under fully warmed-up conditions.

The 75W lubricant containing MoS_2 averaged 5.2 percent increase in fuel economy during warm-up but showed a decrease of 1.3 percent under fully warmed-up conditions. At this stage of the program, a malfunction in the wheel torque sensor prevented torque measurements and subsequent horsepower calculations to compare the 75W with MoS_2 to the base lubricant. However, fuel economy was recorded as previously.

7.2 70°F AMBIENT

During vehicle warm-up, the 75W synthetic lubricant showed an average of 1.2 percent increase in fuel economy (Figure 11) and a slight decrease under fully warmed-up conditions. The horsepower losses were 3.1 hp for the 75W synthetic vs 4.6 hp

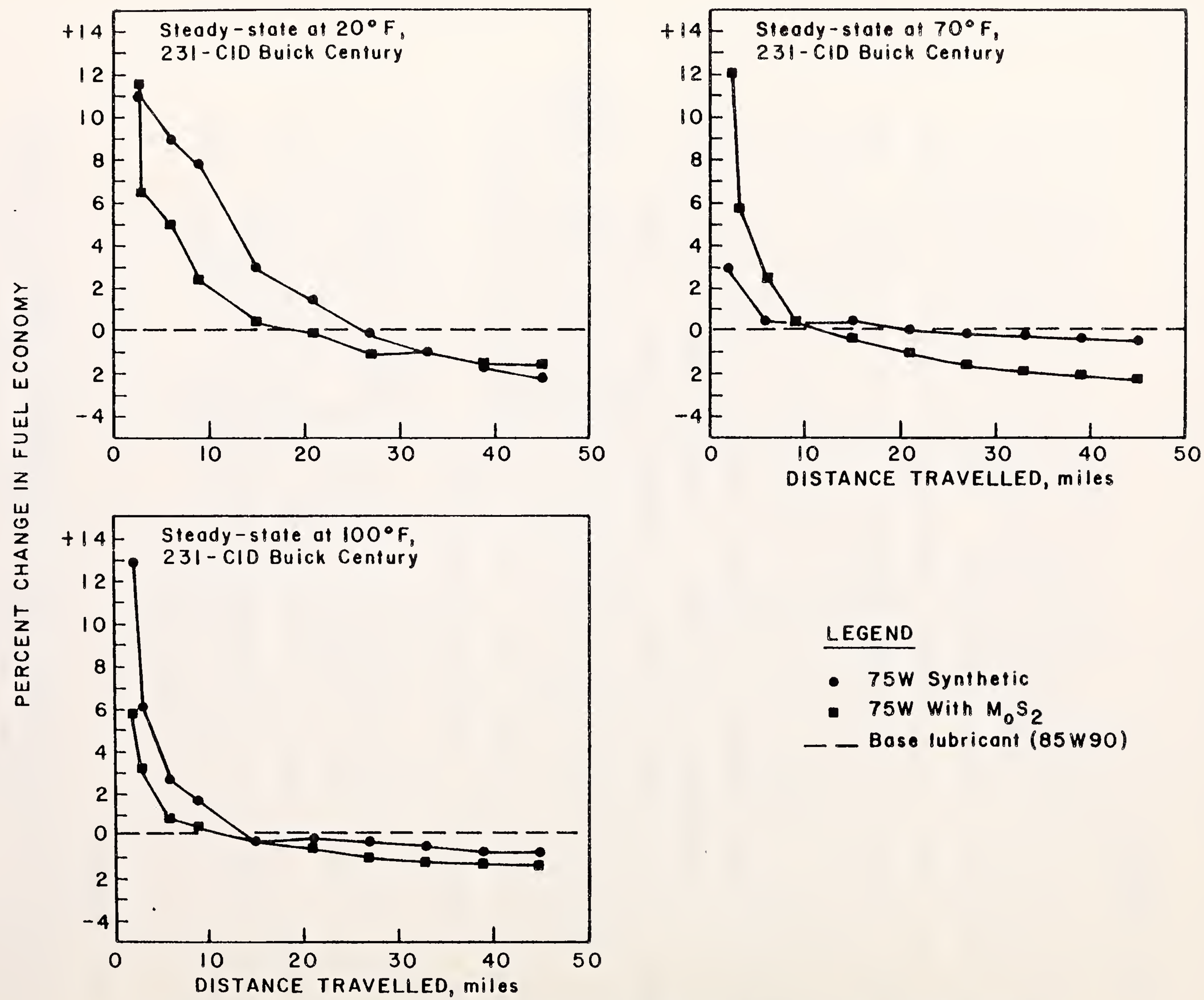


FIGURE 11. COMPARISON OF FUEL ECONOMY WITH GEAR LUBRICANTS IN STEADY-STATE OPERATION

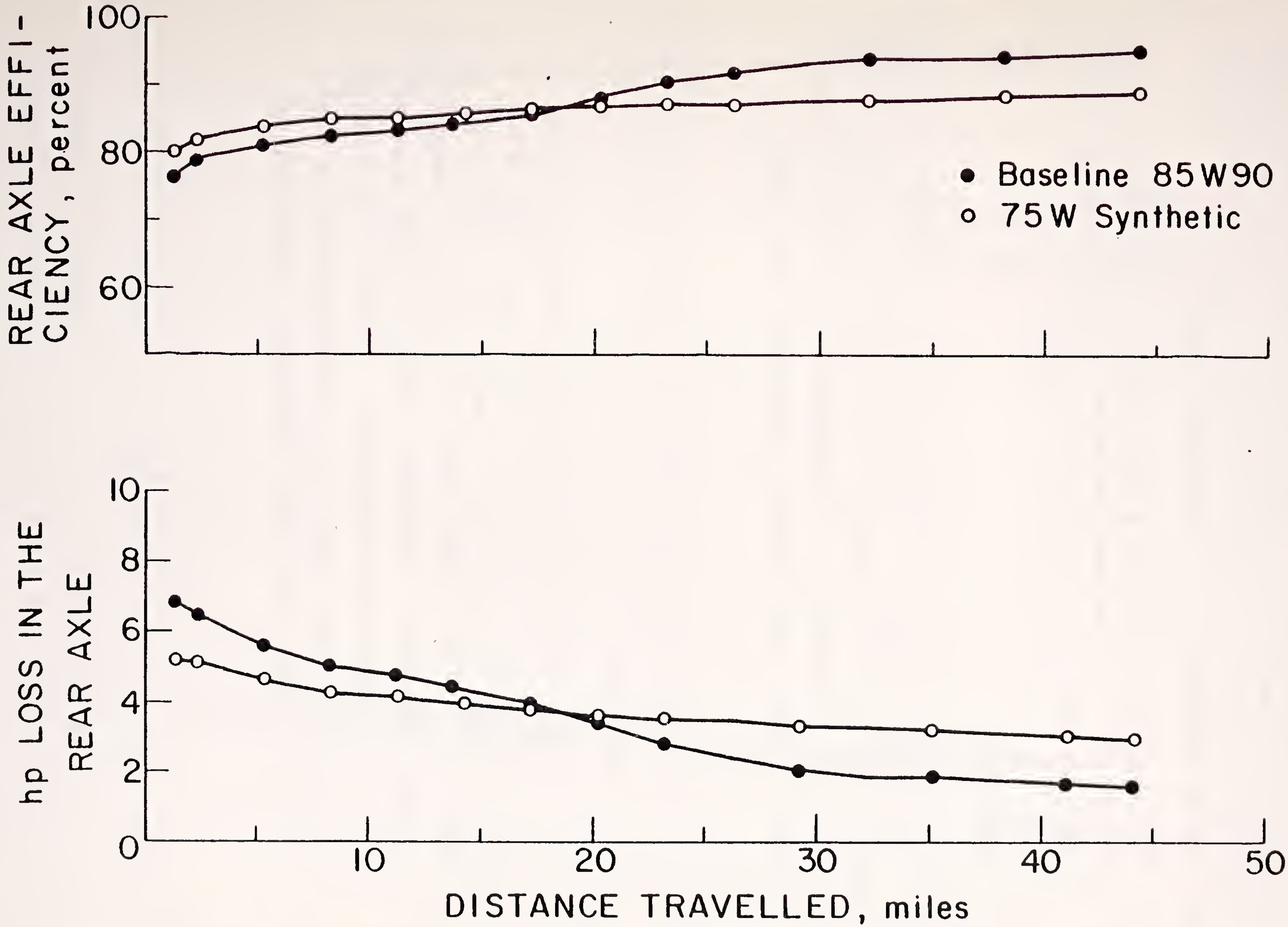


FIGURE 12. COMPARISON OF AXLE EFFICIENCY AND HORSEPOWER LOSS IN STEADY-STATE OPERATION, 20°F, 231-CID BUICK

for the 85W90 during warm-up. Under fully warmed-up conditions, the losses were 2.3 and 2.2 hp, respectively. The corresponding axle efficiencies were 87.4 and 90.3 percent for the 75W synthetic, 82.8 and 91.6 percent for the 85W90 (Figure 13).

The 75W lubricant containing MoS_2 showed an average of 3.5 percent increase and 1.4 percent decrease in fuel economy from warm-up to fully warmed-up conditions (Figure 11).

7.3 100°F AMBIENT

Again, as in the cases of the 20° and 70°F ambients, the 75W synthetic lubricant averaged 4.1 percent increase in fuel economy during warm-up and a slight decrease under fully warmed-up conditions (Figure 11). However, the horsepower losses in the differential were minimal, and the axle efficiency was higher (Figure 14).

The 75W lubricant with MoS_2 showed an average of 1.7 percent increase and 0.9 percent decrease in fuel economy from warm-up to fully warmed-up conditions (Figure 11).

The above results suggest the apparent benefits of low-viscosity gear lubricants in short-trip operation from cold start, whether they are synthetic or treated with friction-reducing additives. On the other hand, these results suggest the benefits of higher viscosity lubricants which provide wear protection under fully warmed-up conditions. However, it remains to be seen whether low-viscosity gear oils provide adequate wear protection under various ambient and operating conditions.

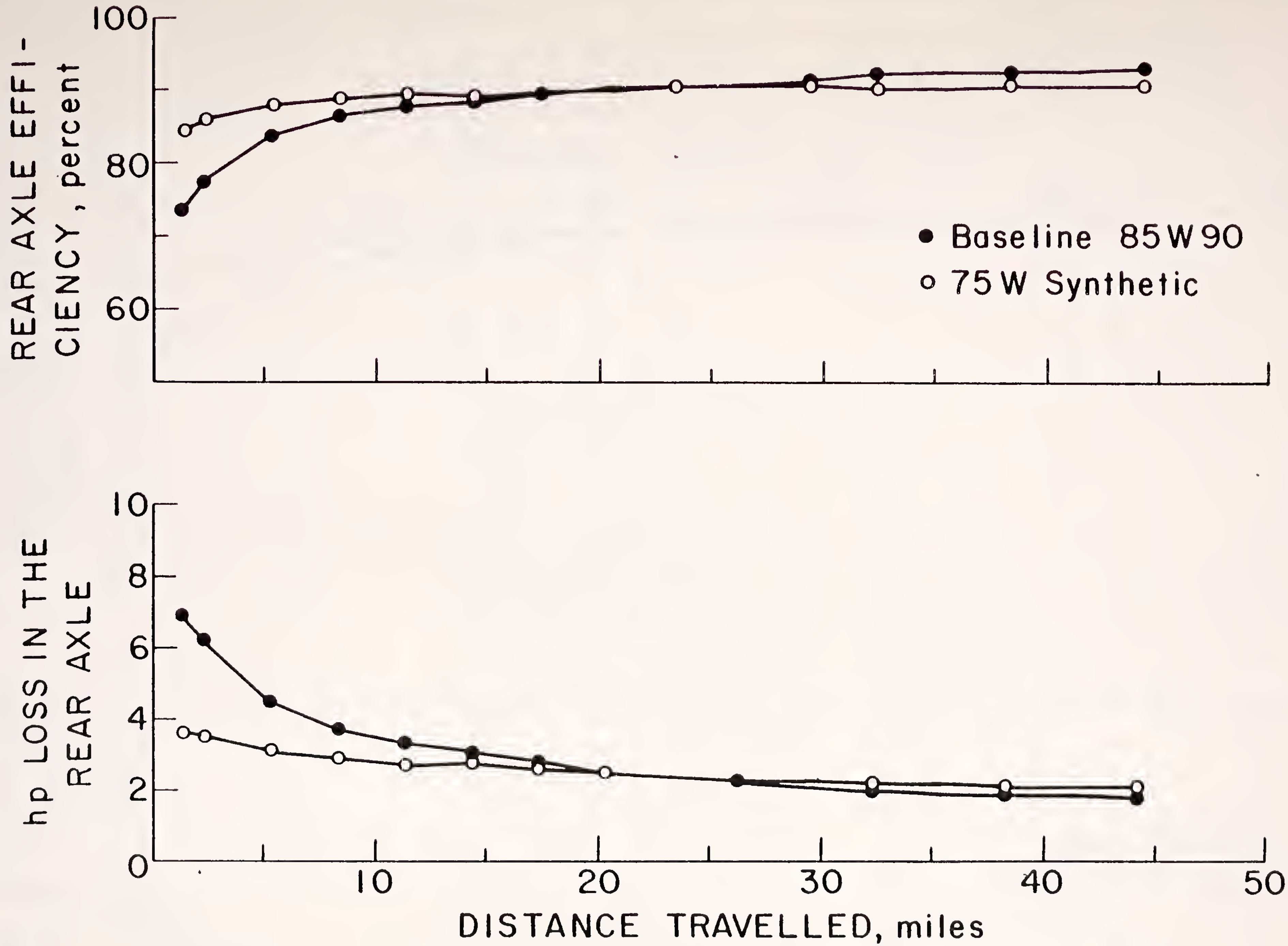


FIGURE 13. COMPARISON OF AXLE EFFICIENCY AND HORSEPOWER LOSS IN STEADY-STATE OPERATION, 70°F, 231-CID BUICK

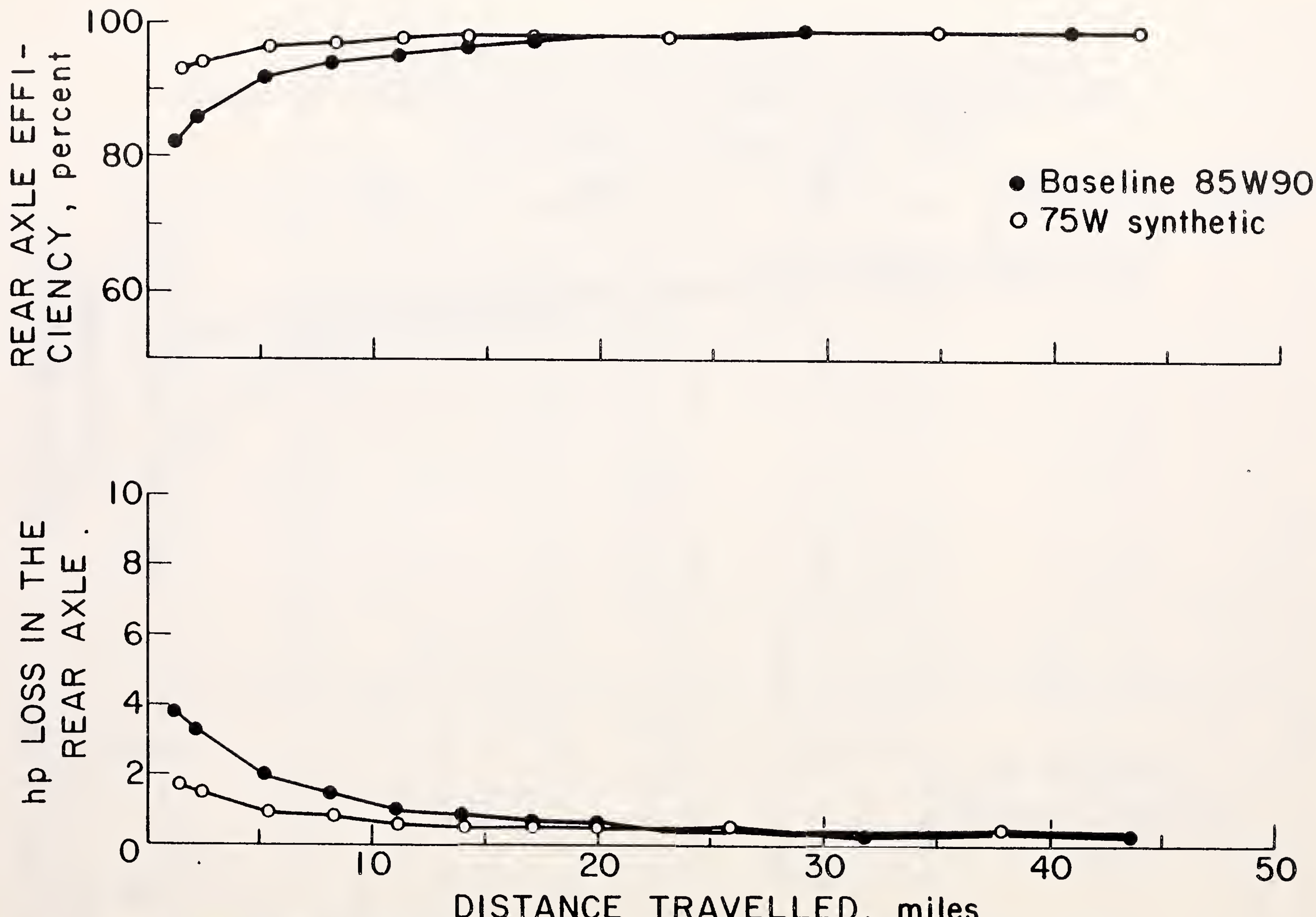


FIGURE 14. COMPARISON OF AXLE EFFICIENCY AND HORSEPOWER LOSS IN STEADY-STATE OPERATION, 100°F, 231-CID BUICK

7.4 GEAR OIL TEMPERATURE EFFECT

Figure 15 shows the oil temperatures for the three gear lubricants at the three ambient temperatures in steady state operation. The 75W synthetic and 75W containing MoS_2 ran consistently at lower temperatures than the baseline 85W90. The temperature difference ranged from 20° to 25°F. As one might expect, this temperature difference could be interpreted as less heat dissipation in the differential, and, therefore, reduced horsepower losses and higher efficiencies. This was true at all three ambient temperatures only during warm-up conditions with the 75W lubricants. However, under fully warmed-up conditions, the 85W90 lubricant showed lower horsepower losses and higher axle efficiencies.

Apparently, this temperature difference among the three gear lubricants is related to two factors: (1) the inherent properties of the synthetic lubricants, which show superior temperature-viscosity characteristics, particularly at extreme temperatures, than straight mineral gear lubricants; particularly at extreme temperatures, than straight mineral gear lubricants; and (2) the addition of dispersed solid lubricants to the base stock, which increased the viscosity index (Table 2) and, therefore, improved the viscosity-temperature characteristics of the final blend.

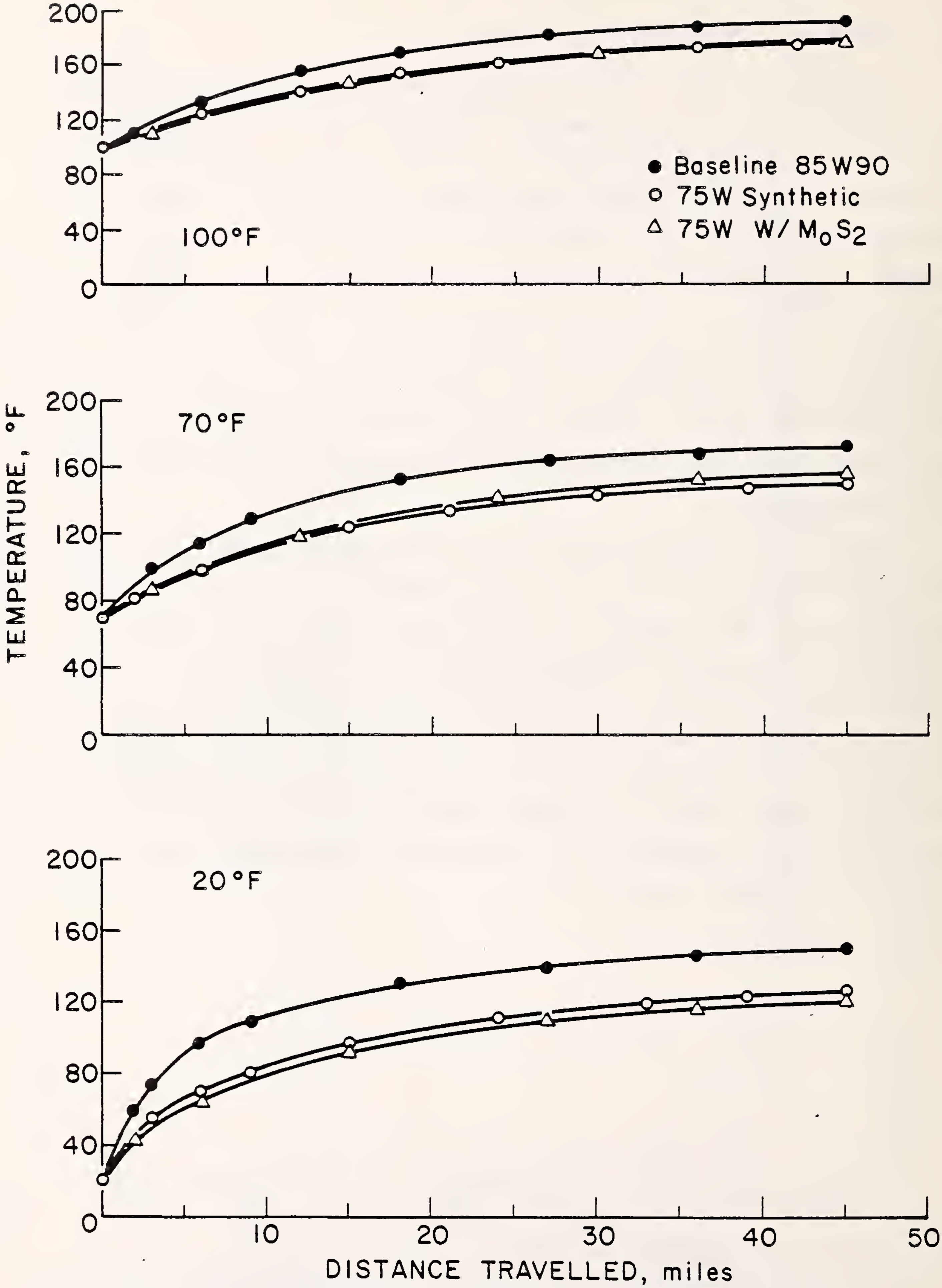


FIGURE 15. DIFFERENTIAL GEAR OIL TEMPERATURES IN STEADY-STATE OPERATION, 231-CID BUICK

8. SUMMARY OF RESULTS

Generally, in the FTP, the increase in fuel economy with a 5W20 and a 10W40 synthetic crankcase lubricant ranged from 1.0 to 4.0 percent when compared to a base 10W40 lubricant. The increase in fuel economy with a graphite-containing 10W40 lubricant ranged from 0 to 6.0 percent, depending on the ambient temperature.

In steady state operation at 60 mph, the fuel economy associated with the synthetic and graphite lubricants increased at all ambient temperatures during vehicle warm-up from cold start, but the results varied from one vehicle to another. Under fully warmed-up conditions, the effect was much smaller.

In steady state operation at 60 mph, a 75W synthetic gear lubricant and a 75W gear lubricant containing MoS_2 showed increased fuel economy, lower horsepower losses, and higher axle efficiencies when compared to an 85W90 base lubricant. These benefits were obtained only during vehicle warm-up following cold start. Under fully warmed-up conditions, there was little or no change in fuel economy.

Standardized quantification of the influence of lubricants on automotive fuel economy requires a comprehensive and uniform test procedure; such a procedure is being developed by the American Society for Testing and Materials.

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APPENDIX A

TABLE A-1. FUEL ECONOMY RESULTS[1]

Test vehicle	Driving schedule	Ambient temp., °F	Fuel economy, mpg Mean	Fuel economy, mpg Std. dev.	Coeff. of variation, pct
10W40 BASE OIL					
Buick...	EPA City	20	14.49	0.10	0.69
		70	17.33	0.08	0.45
		100	17.86	0.32	1.77
	EPA Highway	20	22.17	0.29	1.31
		70	23.15	0.22	0.95
		100	22.87	0.40	1.76
Ford....	EPA City	20	16.71	0.31	1.86
		70	18.71	0.13	0.68
		100	19.85	0.40	1.99
	EPA Highway	20	26.58	0.23	0.88
		70	27.53	0.33	1.18
		100	28.50	0.25	0.87
5W20 SYNTHETIC					
Buick...	EPA City	20	14.75	0.28	1.92
		70	17.56	0.31	1.77
		100	17.95	0.39	2.17
	EPA Highway	20	22.63	0.21	0.94
		70	23.69	0.20	0.84
		100	23.30	0.54	2.31
Ford....	EPA City	20	17.01	0.19	1.12
		70	18.52	0.41	2.21
		100	20.02	0.55	2.74
	EPA Highway	20	26.29	0.18	0.67
		70	27.45	0.48	1.76
		100	28.70	0.40	1.40

[1]Based on triplicate tests.

TABLE A-1. FUEL ECONOMY RESULTS[1] (Cont.)

Test vehicle	Driving schedule	Ambient temp., °F	Fuel economy, mpg		Coeff. of variation, pct
			Mean	Std. dev.	
10W40 SYNTHETIC					
Buick...	EPA City	20	14.03	0.07	0.50
		70	17.56	0.04	0.20
		100	18.29	0.04	0.23
	EPA Highway	20	22.31	0	0
		70	23.79	0.08	0.32
		100	23.54	0.04	0.17
Ford....	EPA City	20	16.63	0.33	2
		70	18.99	0.35	1.86
		100	19.87	0.13	0.66
	EPA Highway	20	25.83	0.54	2.11
		70	27.40	0.37	1.36
		100	27.80	0.13	0.47
10W40 w/GRAPHITE					
Buick...	EPA City	20	14.57	0.32	2.22
		70	17.40	0.16	0.93
		100	18.13	0.22	1.20
	EPA Highway	20	22.86	0.14	0.63
		70	23.52	0.28	1.20
		100	23.71	0.70	2.95
Ford....	EPA City	20	16.61	0.17	1.02
		70	19.38	0.01	0.07
		100	20.19	0.19	0.94
	EPA Highway	20	26.28	0.37	1.43
		70	27.93	0.05	0.22
		100	28.27	0.11	0.39

[1]Based on triplicate tests.

APPENDIX B

TABLE B-1. 60 MPH STEADY-STATE FUEL ECONOMY RESULTS

Test vehicle	Test oil	No. of tests	No. of measurements (1 min. intervals)	Fully warmed-up fuel economy, mpg Mean	Std. dev.	Coeff. of variation, pct
20° F AMBIENT						
Buick...	10W40 base	2	34	18.69	0.03	0.15
	5W20 synthetic	2	34	18.97	0.28	1.45
	10W40 synthetic	3	51	19.18	0.35	1.83
	10W40 graphite	2	34	18.17	0.70	3.40
Ford....	10W40 base	3	51	24.37	0.51	2.11
	5W20 synthetic	3	51	24.17	0.30	1.22
	10W40 synthetic	3	51	24.26	0.32	1.33
	10W40 graphite	2	34	23.83	1.10	4.60
70° F AMBIENT						
Buick...	10W40 base	2	34	20.72	0.05	0.24
	5W20 synthetic	3	51	20.45	0.13	0.66
	10W40 synthetic	3	51	21.05	0.29	1.37
	10W40 graphite	3	51	20.53	0.50	2.45
Ford....	10W40 base	3	51	26.30	0.47	1.78
	5W20 synthetic	3	51	26.09	0.32	1.24
	10W40 synthetic	3	51	24.89	0.67	2.68
	10W40 graphite	2	34	26.39	0.50	1.90
100° F AMBIENT						
Buick...	10W40 base	3	51	19.33	1.08	5.57
	5W20 synthetic	2	34	20.51	0.69	3.38
	10W40 synthetic	3	51	20.32	0.62	3.03
	10W40 graphite	2	34	20.56	0.02	0.10
Ford....	10W40 base	3	51	27.23	0.20	0.73
	5W20 synthetic	3	51	27.13	0.62	2.27
	10W40 synthetic	3	51	25.95	0.95	3.67
	10W40 graphite	2	34	27.73	0.15	0.54

APPENDIX C

TABLE C-1. 60 MPH STEADY-STATE FUEL ECONOMY[1]

Test gear oil	Fully warmed-up fuel economy, mpg		Coeff. of variation, pct
	Mean	Std. dev.	
20° F AMBIENT			
85W90 base......	19.80	0.16	0.79
75W synthetic...	19.81	0.38	1.92
75W + MoS_2......	19.43	0.35	1.82
70° F AMBIENT			
85W90 base......	21.55	0.21	0.95
75W synthetic...	21.57	0.09	0.43
75W + MoS_2......	20.85	0.41	1.97
100° F AMBIENT			
85W90 base......	21.39	0.09	0.43
75W synthetic...	21.23	0.01	0.03
75W + MoS_2......	21.30	0.59	2.72

[1]Results based on duplicate tests conducted on the 1978 Buick.

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